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## Optimized damper control of pressure and airflow in ventilation systems

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**KEYWORDS:** ventilation system, HVAC control strategy, variable air volume, static pressure reset, modelling, Simulink, energy savings.

### **SUMMARY:**

*Conventional control strategies in variable air volume (VAV) ventilation systems do not take fully into advantage the potential energy savings since the system operation is based on maintaining a constant static pressure (CSP) set point in the main duct irrespective of the actual pressure demand. The static pressure reset (SPR) control strategy can optimize the operation of the supply air fans by adjusting the pressure set point to be just enough to deliver the required airflow to the most critical zone.*

*This study investigated the operation and energy savings potential of an SPR control algorithm by using the Simulink programming tool which is add-on software to MATLAB mathematical programming language. A model of a VAV ventilation system was created in Simulink based on the International Building Physics Toolbox (IBPT); the IBPT thermal zone was remodelled in order to calculate dynamically the airflow demand according to the zone air temperature. The performance of the Simulink model was evaluated based on the experimental setup of the ventilation system. The SPR control method established stable system operation and was proven efficient to maintain comfortable space conditions while reducing by 14 % the fan energy used in a typical working day.*

## **1. Introduction**

In traditional control of variable air volume (VAV) systems the terminal boxes and the air handling unit (AHU) are operated independently without integration. The common practice is to control the AHU to a constant static pressure (CSP) set point corresponding to the pressure rise required under the design full load condition (Wei et al 2004). However in this way the AHU is regulated irrespective of the actual pressure demand. This is because under part load condition the fan is providing excessive static pressure (Wei et al 2004; Federspiel et al 2005; Liu et al 1997) which is dissipated by increasing the airflow resistance of the air distribution network via throttling at the terminal boxes. As a result significant fan power is wasted in mechanical energy losses. By integrating the control of terminal boxes into the building management system (BMS) it is possible to implement the static pressure reset (SPR) control strategy. This method regulates the AHU in real time according to feedback from several individual zones. In this case the fan generates the pressure required in order to satisfy the space conditions in the most critical zone while maintaining the airflow resistance of the distribution network at a minimum (Wang et al 1998). Consequently the fan pressure rise and thus the fan power is reduced. The control method of trim and respond based on zone pressure request alarms is the most efficient SPR strategy since it is more stable, flexible and it minimizes the impact of rogue zones (Taylor 2007). The objective of this paper is to investigate the operation and the potential energy savings of the trim and respond SPR control strategy. A mathematical model of a conventional VAV ventilation system is developed in Simulink (Simulink 2013) and the model is expanded by implementing the optimized SPR control algorithm. The Simulink model operation is validated by experiments performed on the full-scale test system.

## 2. The Simulink model

The model of the VAV ventilation system was created in the graphical environment of Simulink in Matlab (Matlab 2013) and it was built based on the blocks of the international building physics toolbox (IBPT). The IBPT toolbox (IBPT 2012) is a library of blocks added on Simulink, specially constructed for the thermal analysis in building physics. The construction blocks (external and internal surfaces, windows) provide detailed calculations of the thermal state of every subcomponent in the structure according to the surrounding conditions to which it is exposed. The thermal condition of the zone is calculated according to the heat gained through the building envelope, the systems used for heating, ventilation and air conditioning, the internal gains occurring in the zone and the weather data corresponding to a certain location. The default IBPT blocks for the internal gains and the ventilation system were rebuilt in order to fit the operation needs of the VAV ventilation system. FIG 1 illustrates the Simulink model which consists of three IBPT thermal zones.

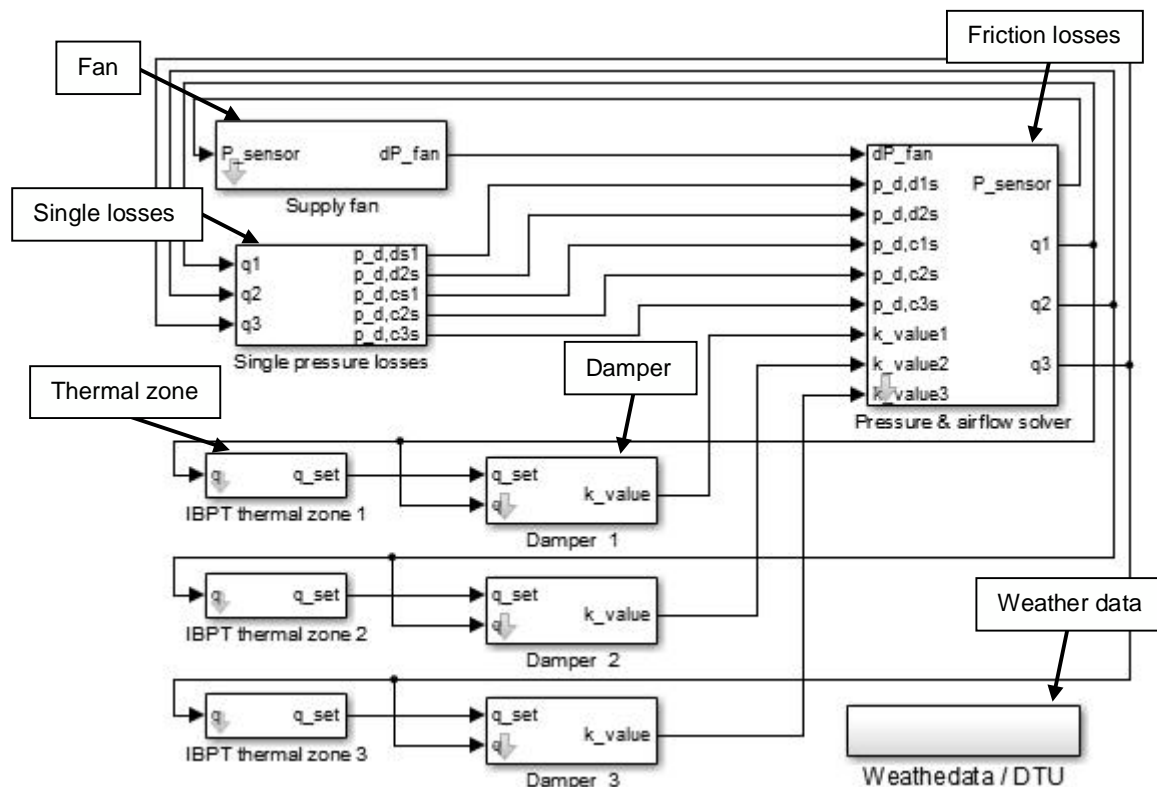


FIG 1. The Simulink model of the VAV ventilation system.

### 2.1 The internal gains

The IBPT internal gains block was configured to include an hourly load schedule of the ventilated zone. The modified block calculates the heat gains based on user defined profiles considering occupants, equipment and lighting use in the zone.

### 2.2 The ventilation system

The IBPT ventilation system block was remodelled to calculate dynamically the airflow demand ( $q_{dem}$ ) according to the zone air temperature ( $T_a$ ); the strategy is implemented with the ramp functions shown in FIG 2. The user defined data are the minimum ( $q_{set,min}$ ) and maximum ( $q_{set,max}$ ) airflow set point required in order to maintain a comfortable temperature range ( $T_{set,min}$ ,  $T_{set,max}$ ) in the zone.

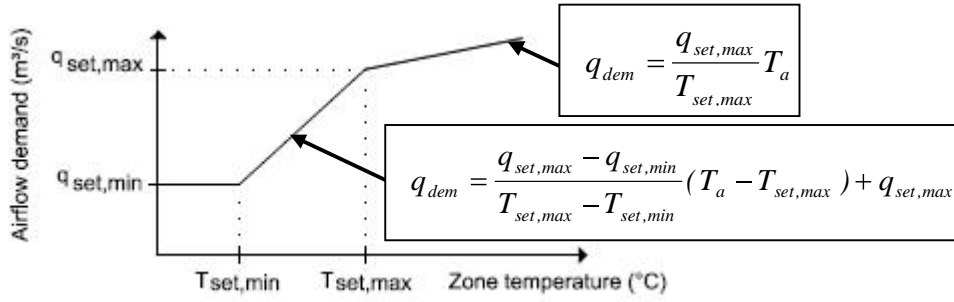


FIG 2. The dynamic calculation of the airflow demand according to the zone air temperature.

### 2.3 The fan

The fan operation is regulated according to the tracking error determined as difference between the CSP set point and the duct pressure at the sensor position; the block diagram is shown in FIG 3.

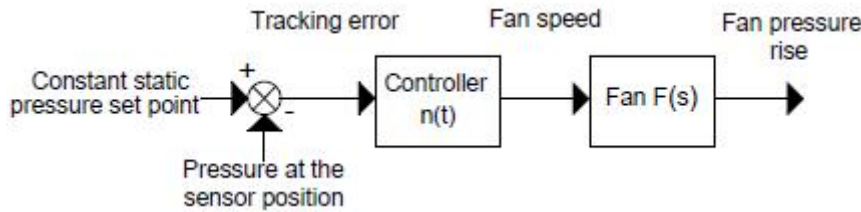


FIG 3. The fan operation principle.

The fan block receives the signal generated by the controller which corresponds to the fan speed. This control signal is used to calculate the new plant output, the fan pressure rise. The fan characteristics (F) are represented by using the first-order linear time-invariant (LTI) system (Franklin et al 1993) given in equation 1.

$$F(s) = \frac{k_f}{Ts + 1} \quad (1)$$

Where  $s$  representation of Laplace transformation (-)  
 $k_f$  the process gain correlating the plant input with the plant output (Pa/rpm)  
 $T$  the time constant is the time required to reach the system a steady state condition (sec)

The controller operation complies with equation 2; the tuning of the proportional integral (PI) controller is performed by using the simple analytic rules proposed by S. Skogestad (2002).

$$n(t) = K_{p,f} e_f(t) + K_{i,f} \int e_f(t) dt \quad (2)$$

Where  $e_f$  the tracking error between the CSP set point and the pressure sensor reading (Pa)  
 $K_{p,f}$  the proportional controller gain, 7.14  
 $K_{i,f}$  the integral controller gain, 17.85

### 2.4 The damper

The damper operates as shown in FIG 4; the damper position is adjusted based on the tracking error determined as difference between the zone airflow demand (see FIG 2) and the airflow provided to the zone. In the control process in FIG 4 the plant block is representing the damper system that receives the controller signal which corresponds to the damper position. The plant output, the resistance coefficient, is calculated accordingly. The second-order LTI system (Franklin et al 1993) presented in equation 3 approximates the operation of a typical damper (D).

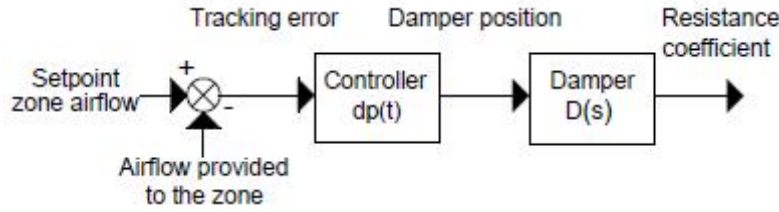


FIG 4. The damper operation principle.

$$D(s) = \frac{k_d \omega_n}{s^2 + 2\zeta \omega_n s + \omega_n} \quad (3)$$

Where  $k_d$  the process gain correlating the plant input with the plant output ( $\text{m}^3/\text{s}/\text{Pa}\%$ )  
 $\omega_n$  the natural frequency relevant to the speed response of the system, assumed 10 rad/s  
 $\zeta$  the damping ratio relevant to the oscillation mode of the system, assumed 1

The controller operation is applied according to equation 4; the tuning gains of the PI controller are set based on typical product values.

$$dp(t) = K_{p,d} e_d(t) + K_{i,d} \int e_d(t) dt \quad (4)$$

Where  $e_d$  the tracking error between the demanded and the delivered airflow in the zone ( $\text{m}^3/\text{s}$ )  
 $K_{p,f}$  the proportional controller gain, 1  
 $K_{i,f}$  the integral controller gain, 10

## 2.5 The pressure and airflow solver

The friction and single pressure losses blocks illustrated in FIG 1 implement the hydraulic calculation of the VAV ventilation system according to the duct design shown in FIG 5. The unknown pressure and airflow conditions are determined by setting up a system of equations expressing the pressure losses occurring in every component of the system. The hydraulic calculation determines the pressure demand (P) at the beginning and end of every component as well as the airflows (q) delivered to the different zones (see FIG 5). The system of equations cannot be solved analytically; therefore the Newton-Raphson numerical method is used instead.

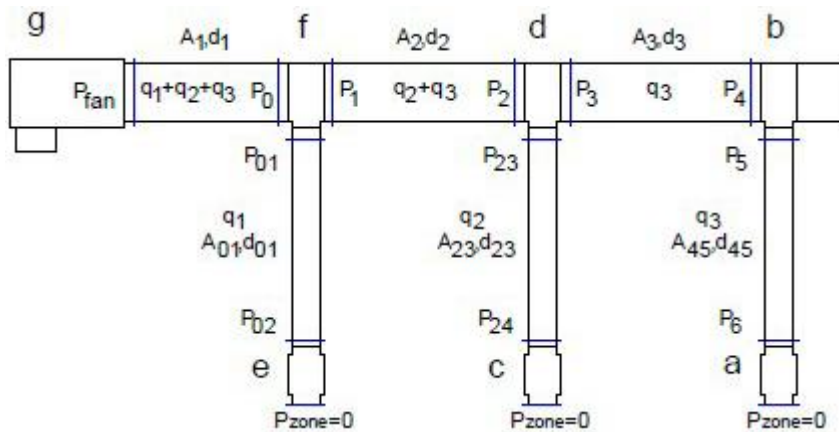


FIG 5. The duct design in the pressure and airflow solver block.

In a piece of duct the pressure losses due to friction are calculated according to the Darcy-Weisbach equation (White 1998) given in equation 5. The Darcy friction factor is obtained by the Swamme-Jain equation (Swamme et al 1976), which is an approximation of the implicit Colebrook-White equation (see equation 6).

$$\Delta P_{fr} = f_D \frac{L\rho}{d2} u_{air}^2 \quad (5)$$

Where  $f_D$  the Darcy factor (-)  
 $L$  the length of the duct (m)  
 $d$  the diameter of the duct (m)

$$f_D = \frac{0.25}{\left( \log_{10} \left( \frac{5.74}{Re^{0.9}} + \frac{\varepsilon}{3.7d} \right) \right)^2} \quad (6)$$

Where  $Re$  the Reynolds number (-)  
 $\varepsilon$  the roughness height, for thin plate ducts is equal to  $0.15 \times 10^{-3}$  m

The pressure losses due to connections and fittings are calculated according to equation 7.

$$\Delta P_{sing} = \frac{\zeta_r u_{air} \rho}{2} \quad (7)$$

Where  $\zeta_r$  the resistance coefficient (-)  
 $u_{air}$  the mean velocity of airflow (m/s)  
 $\rho$  the density of air,  $1.204 \text{ kg/m}^3$

The pressure losses introduced by the damper component are approximated based on equation 8.

$$\Delta P_{sing} = \left( \frac{q_{del}}{k_{value}} \right)^2 \quad (8)$$

Where  $q_{del}$  the airflow delivered to the zone ( $\text{m}^3/\text{s}$ )  
 $k_{value}$  the damper resistance coefficient ( $\text{m}^3/\text{sPa}$ )

## 2.6 The static pressure reset algorithm

In order to implement the SPR control method of trim and respond based on zone pressure request alarms, one more block was added to the Simulink model presented in FIG 1. The operation principle of the applied control logic can be seen in FIG 6.

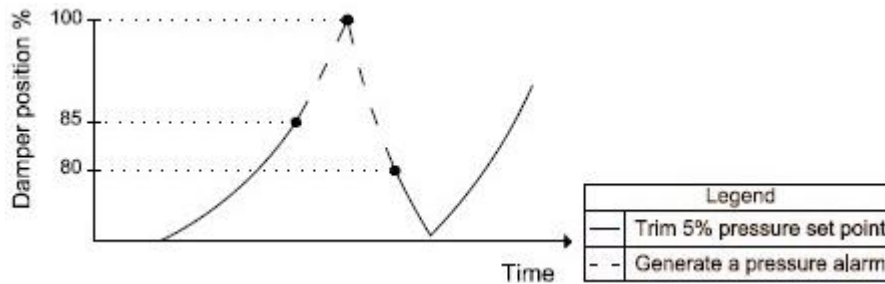


FIG 6. The control logic of the trim and respond static pressure reset method.

Every damper of the VAV system transmits an alarm signal when its position exceeds 85 % opening; the zone keeps generating a pressure request until the damper closes to a position of 80 % opening. The pressure requests from all zones are summed and when at least two zones give an alarm the fan pressure set point is reset 10 % upwards of the pressure demand at the sensor position. In the opposite case it is reset 5 % downward. The SPR is performed within a specific pressure range; the upper limit is set equal to the CSP set point while the lower SPR limit is determined according to the pressure demand ensuring precise damper operation. The SPR loop resets the pressure set point every 90 sec

and the fan adjusts to the new pressure set point.

### 3. The experimental setup

The performance of the Simulink model was validated on a full scale experimental setup; the experimental setup arrangement was identical to the duct design given in FIG 5 where the distribution duct had a diameter of 315 mm and the connection ducts a diameter of 160 mm. The setup consisted of three LeanVent DropDamper LERX and a box fan (Exhausto BESF1804-1EC). The VAV system was evaluated both with the CSP and the SPR control strategy; the two methods were modelled in LabVIEW (LabVIEW 2013). The validation was performed by providing the Simulink model and the experimental setup with the same airflow demand data; two different airflow demand profiles were used for testing the model performance with each control strategy (see FIG 7). Ventilation zone 3 behaved like a rogue zone with the SPR method because the lower limit of the pressure range, within which the fan operation set point reset, was insufficient for delivering the required pressure.

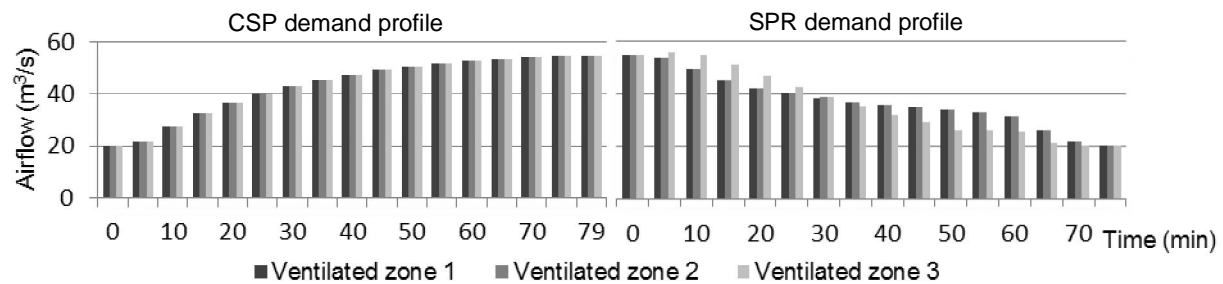


FIG 7. Airflow demand profiles with the CSP and the SPR control method, respectively.

## 4. Results

### 4.1 Simulink model validation

The graphs in FIG 8 compare the performance of the Simulink model and the experimental setup of the VAV ventilation system when the fan was controlled with the CSP and the SPR control method, respectively.

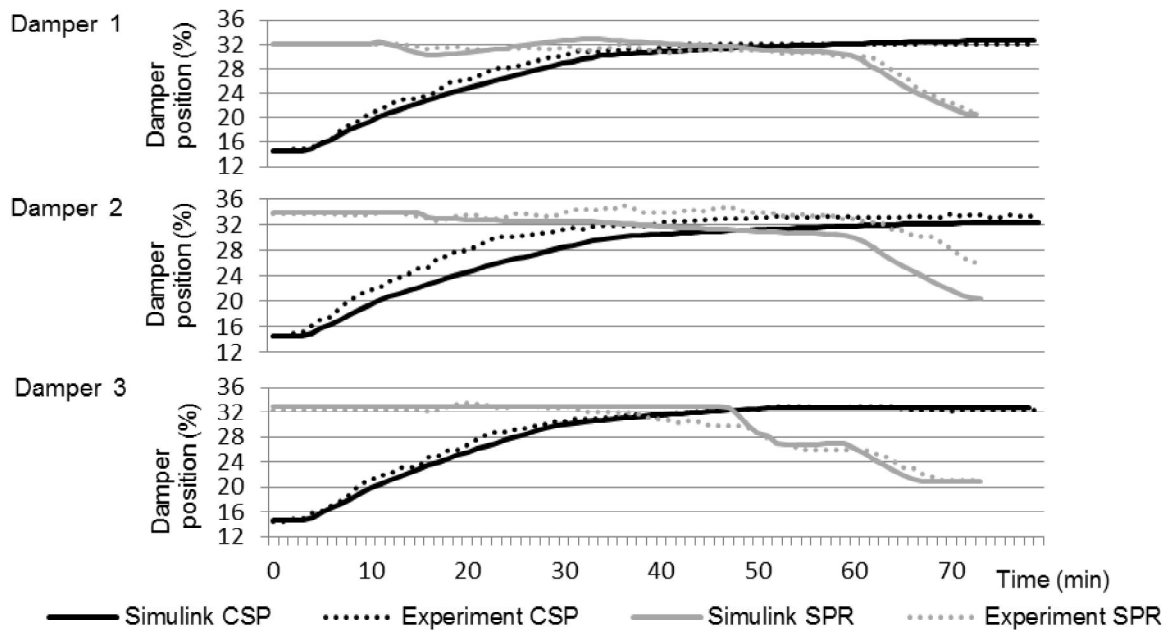


FIG 8. Response curves of the dampers with the CSP and the SPR control method, respectively.

According to the obtained results the Simulink model with both control strategies approximated satisfactory the actual operation conditions since the dampers followed a similar response trend. The largest deviation between the Simulink model and the experimental setup was obtained from damper 2 when the VAV system was regulated with the SPR control method. As presented in the second graph in FIG 8 the deviation was below 10 %, however in the last minutes it increased to 22 %. This occurred because the damper modelling represented insufficiently the actual speed response of the Dropdamper. The mathematical model of the damper should be tuned to coincide with the experimental data. The speed of the damper model is relative to the natural frequency parameter involved in the mathematical expression describing its response (see equation 3). In this case the speed of the damper was assumed; in order to accelerate the response, trial and error simulations have to be performed increasing the natural frequency.

## 4.2 Energy savings from optimized damper control

The energy savings potential of the SPR control strategy is determined based on the fan power used when controlling the ventilation system with the CSP and the SPR method, respectively. Considering that the fan efficiency varies according to the different pressure and airflow conditions that the fan is operating with, the fan efficiency was approximated from producer table values by using average hourly values of airflow and pressure rise delivered by the fan. The average hourly values were derived based on 24 hour simulation data obtained from the Simulink model of the VAV ventilation system when operated with both control strategies. The fan power was calculated according to equation 9.

$$Power = \frac{\Delta P_{fan} q_{tot}}{n_e} \quad (9)$$

Where  $\Delta P_{fan}$  the fan pressure rise (Pa)  
 $q_{tot}$  the fan airflow (m<sup>3</sup>/s)  
 $n_e$  the fan efficiency (-)

The fan energy used in a typical working day in winter when the occupancy period was set from 8 am to 17 pm with the CSP and the SPR was determined to 151 Wh and 130 Wh, respectively; the energy consumption was reduced approximately by 14 %. The highest energy savings potential of the SPR control method occurred under part airflow conditions as the fan operated with decreased static pressure. Due to the fact that the fan was correctly sized the combination of lower pressure set point and airflow improved fan efficiency and thus the energy savings were further increased. With the CSP control method the same airflow conditions combined with increased static pressure lowered the fan efficiency and as a result the fan operated inefficiently.

## 5. Conclusions

The results presented in this paper draw the following conclusions:

- The first and the second-order LTI systems were proven representative for the response of the fan and the damper, respectively. For the current study we assumed the parameters of the mathematical model of the damper ( $\zeta$ ,  $\omega_n$ ). The value of the natural frequency parameter turned to be inaccurate for representing the speed response of the Dropdamper as it introduced high deviation to the Simulink model when it was controlled with the SPR control. The damper mathematical model should be tuned to fit the experimental data.
- The SPR control method documented higher energy savings under part airflow conditions where the fan operated with decreased pressure set point.

In practical applications of the SPR control strategy caution should be given when determining the lower limit of the pressure range within which the variable fan operation set point is established. This



parameter is critical because in case that the minimum fan pressure rise is insufficient to satisfy the system pressure demand, the far located zones will act as rogue zones. Therefore it is advisable to perform airflow measurements in order to ensure that the set point selected is appropriate for the precise operation of the dampers. In order to maximize the energy savings potential of the SPR control method, the fan should be correctly sized; in this case the fan efficiency improves when the fan operates with decreased static pressure.

## 6. Acknowledgements

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